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Solberg, Brian; Andersen, Palle; Stoustrup, Jakob

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# The One-pass Smoke Tube Marine Boiler - Limits of Performance

Brian Solberg<sup>1</sup>, Palle Andersen<sup>2</sup> and Jakob Stoustrup<sup>2</sup>

<sup>1</sup>Aalborg Industries A/S, Aalborg, Denmark, bso@es.aau.dk

<sup>2</sup>Aalborg University, Aalborg, Denmark, {pa,jakob}@es.aau.dk

**ABSTRACT:** This paper summarises a number of results gathered over the past few years regarding modelling and control of the one-pass smoke tube marine boiler. The purpose is to communicate our state of knowledge regarding the limits of performance in these processes. The standpoint of the paper is limits with respect to the current standard configured boiler. We present a simple second order control model for the boiler capturing also cross-couplings and disturbance influence. This model is accurate over a large frequency range including the crossover frequencies. Performance limitations imposed by the actuator systems, sensor noise and neglected model dynamics are treated. Also control design guidelines are presented and suggestions for pushing the limits by new equipment presented.

**Keywords:** limits of performance, control model, constraints, flow control, boiler control, sensor noise

## 1 INTRODUCTION

Over the past few years much research has been directed towards the modelling and control of the one-pass smoke tube marine boiler reported in e.g. [1, 2, 3, 4, 5] along with numerous student projects conducted at Aalborg University in co-operation with Aalborg Industries A/S.

In this paper the results gathered from these works will be used to setup control design guidelines and specify limits of performance for the boiler process. The boiler setup is shown in Figure 1. The boiler family concerned in the present work is the oil-fired one-pass smoke tube boiler from Aalborg Industries A/S (AI). The boiler consists of a furnace and flue gas pipes surrounded by water. In the top of the boiler steam is led out and feed water is injected. This boiler differs from other boiler designs in two ways: it is side-fired and the flue gas passes straight through. As an example one of these boilers is designed

for a maximum steam load of  $1800 \frac{\text{kg}}{\text{h}}$  at operating pressure 8 bar. The minimum steam load is obviously  $0 \frac{\text{kg}}{\text{h}}$  whereas the minimum capacity of the burner unit corresponds to a steam flow of approximately  $400 \frac{\text{kg}}{\text{h}}$ .

The initial interest in the one-pass smoke tube boiler was to obtain a control strategy which was able to minimise the fluctuations in the water level without compromising pressure performance in such a way that the physical geometry of the boiler could be reduced. It was the conviction that such initiatives would require accurate detailed nonlinear first principle models of the boiler and a controller design taking into account the multiple input multiple output and nonlinear characteristics of the process. However, in this paper we will more or less argue the opposite. If the boiler dimensions are to be minimised and hence the process pushed to the limits it is important to know what

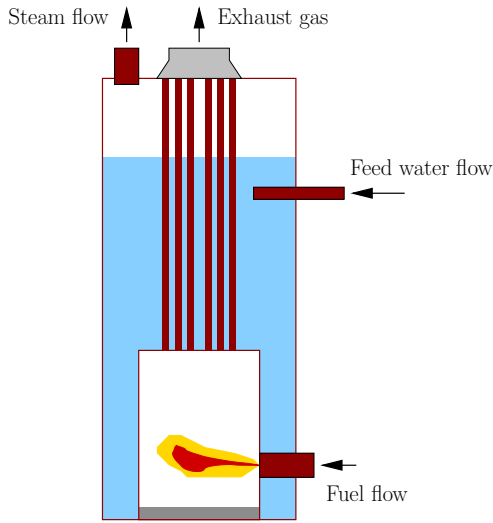


Figure 1: Principle of the oil-fired one-pass smoke tube marine boiler.

set these limits. Likewise studying these limits might help engineers manually tuning controller during commissioning. However, it is also obvious that if simple control design guide lines can be put up, the current relatively long time spent by the engineer tuning the controllers can be reduced remarkably.

Much published work on boiler modelling and control is available, [6, 7, 8, 9, 10]. Both models based on first principles of varying complexity and models based on system identification techniques to specify a black box model based on e.g. linear parametric models have been proposed. Setting up control guide lines can be done using any of the existing control design techniques as: Linear Quadratic Control, [11], Model predictive control [12, 13], Robust control [14, 15] and PID control [16]. Regarding limits of performance tools for analysing linear systems can be found in almost any text book on control theory e.g. [17, 18]. However, for nonlinear systems there are less systematic analysis procedures available.

The focus is on nonlinear model reduction to create a simple second order model including cross terms and disturbance influences. The important nonlinearities will be shown to persist at the input of the plant due to the actuator systems. Regarding performance the con-

straint on the actuator absolute values together with sensor noise will be shown to be the limiting factor. Simple controller design guidelines are presented. For these controllers to be easy to understand and tune by any service engineer the control theory used is based on classical PID control.

The paper is organised as follows. Firstly the simple nonlinear model is described along with simple linearised versions. Secondly the limits of performance are discussed, treating the actuator systems, shrink-and-swell, disturbances, nonlinearities, interaction measurements noise and constraints. In the subsequent section control design guidelines for the actuator systems and the boiler are put up. Lastly concluding remarks are presented.

## 2 MODEL

Here we briefly discuss the nonlinear model of the boiler with one purpose only: how to find a suitable model for controller design. Many dynamical models of varying complexity for the drum boiler have been proposed in the literature – see e.g. [6, 19, 2, 4, 20]. However, already in [21] it was pointed out that a high order linear model was not necessary for describing the dynamics of the boiler important to controller design and further step response analysis showed good agreement between responses from a nonlinear model and linear model. The simple control model does not account for precise stationary gains and further does not provide information on many internal variable. However, when constraints are not present for the internal variables these thing are not important to the control. In particular, a controller will usually include integral action which among others account for model stationary gain mismatches.

Studies have shown that both the flue gas part (furnace and convection tubes) and the metal separating the water/steam part from the flue gas have considerably faster dynamics than the desired closed loop bandwidth with time constants  $< 2$  s. Due to this fact the power delivered to the

water/steam part is modelled as:

$$Q = \eta \dot{m}_{fu} \quad (1)$$

where  $\eta$  is a constant describing a combination of energy released in the combustion plus furnace and convection tubes heat transfer efficiency.  $\eta$  is in fact a function of the burner load and water level in the boiler drum, but for control purposes it is sufficiently accurate to consider  $\eta$  constant. First of all modelling the dependency on the water level was shown in [1] to give rise to some special low frequency phenomena. This was seen as a zero in the origin from fuel flow to pressure and the integrator from feed water to water level was moved slightly into the left half plane. These phenomena are seen at frequencies far below the interesting bandwidth and since water level will always be controlled, which removes the zero in the origin, there is no further need to include this in a control model. Also it turns out that in the boiler family treated here  $\eta$  is approximately invariant to the burner load.

The model of the water/steam part has the purpose of describing the steam pressure in the boiler  $p_s$  and the water level  $L_w$ . The modelling is complicated by the shrink-and-swell phenomenon due to steam load changes which is caused by the distribution of steam bubbles under the water surface.

The total volume of water and steam in the boiler is given as:  $V_t = V_w + V_s + V_b$ , where  $V_w$  is the water volume,  $V_s$  is the volume of the steam space above the water surface and  $V_b$  is the volume of the steam bubbles below the water surface.

To capture the dynamics of the water/steam part the total mass and energy balances are considered. The total mass balance for the water/steam part is:

$$\frac{d}{dt}(\varrho_s(V_t - V_w) + \varrho_w V_w) = \dot{m}_{fw} - \dot{m}_s \quad (2)$$

and the energy balance is:

$$\begin{aligned} \frac{d}{dt} \left[ \varrho_w V_w h_w + \varrho_s (V_t - V_w) h_s + \right. \\ \left. - V_t p_s + \varrho_m V_m c_{p,m} T_s \right] = \\ Q + h_{fw} \dot{m}_{fw} - h_s \dot{m}_s \end{aligned} \quad (3)$$

where  $\dot{m}_{fw}$  is the feed water flow,  $\dot{m}_s$  is the steam flow,  $\varrho$  is density,  $h$  is enthalpy and  $T$  is temperature,  $c_p$  is specific heat capacity and subscript  $m$  stands for metal. It should be noticed that energy accumulated in metal of the boiler jacket, furnace and convection tubes is included in the balance for the water/steam part.

The mass balance can be written as:

$$\begin{aligned} \left[ (V_t - V_w) \frac{d\varrho_s}{dp_s} + V_w \frac{d\varrho_w}{dp_s} \right] \frac{dp_s}{dt} + \\ + (\varrho_w - \varrho_s) \frac{dV_w}{dt} = \dot{m}_{fw} - \dot{m}_s \end{aligned} \quad (4)$$

and as  $\frac{d\varrho_s}{dp_s}$  is  $\approx 10$  times smaller than  $\frac{d\varrho_w}{dp_s}$  we make the following approximation of (4):

$$V_w \frac{d\varrho_w}{dp_s} \frac{dp_s}{dt} + (\varrho_w - \varrho_s) \frac{dV_w}{dt} \approx \dot{m}_{fw} - \dot{m}_s \quad (5)$$

Now following [6] another simple expression for the pressure can be derived. Multiplying (2) by  $h_w$  and subtracting the result from (3) gives:

$$\begin{aligned} \left[ h_c (V_t - V_w) \frac{d\varrho_s}{dp_s} + \varrho_w V_w \frac{dh_w}{dp_s} - V_t + \right. \\ \left. + \varrho_s (V_t - V_w) \frac{dh_s}{dp_s} + \varrho_m V_m c_{p,m} \frac{dT_s}{dp_s} \right] \frac{dp_s}{dt} + \\ - h_c \varrho_s \frac{dV_w}{dt} = Q - (h_w - h_{fw}) \dot{m}_{fw} - h_c \dot{m}_s \end{aligned} \quad (6)$$

where  $h_c = h_s - h_w$  is the vaporisation enthalpy. (5) could be inserted in (6). However, the ratio  $\frac{\varrho_s}{\varrho_w - \varrho_s} = 0.0047$  is small for which reason we neglect the  $\frac{dV_w}{dt}$  term in (6). The term multiplying  $\frac{dp_s}{dt}$  has large differences in numerical size and a good approximation of the pressure dynamics is due to the large water volume in the boiler given by:

$$\frac{dp_s}{dt} \approx \frac{Q - (h_w - h_{fw}) \dot{m}_{fw} - h_c \dot{m}_s}{\varrho_w V_w \frac{dh_w}{dp_s}} \quad (7)$$

Equations (5) and (7) above only express the pressure and the water volume in the boiler. As the water level of interest in the control problem is given as:  $L_w = (V_w + V_b - V_o)/A_{ws}$ , another equation is

needed for describing the volume of steam bubbles  $V_b$  in the water (the water level is measured from the furnace top and  $V_o$  is the volume surrounding the furnace and  $A_{ws}$  is the water surface area). Many proposals describing the distribution of steam bubbles under the water surface have been made – see e.g. [6, 2, 20, 10]. Most of these are based on assumptions and all end up including empirical constants to be estimated to fit the model to process data. In [19] an approach was taken to model the boiler as a collection of linear models in which a non-minimum phase zero is easily inserted.

The difficult part of the modelling is to describe the amount of steam escaping the water surface,  $\dot{m}_{b \rightarrow s}$ . Here we take the approach of [20] which is similar to the expression in [6]:

$$\dot{m}_{b \rightarrow s} = \frac{\varrho_s}{T_d} V_b \quad (8)$$

where  $\dot{m}_{b \rightarrow s}$  is expressed as a function of the steam bubble volume and density of the steam, the constant  $T_d$  expresses the average rise time of bubbles in the water. This flow can be used to set up a mass balance for the water and steam below the water surface. However, the dynamics of this extra mode is very fast with a time constant of about 1 s. Therefore it is reasonable to assume a stationary relationship between the steam load and bubble volume as:

$$V_b = \frac{T_d}{\varrho_s} \dot{m}_s \quad (9)$$

This equation introduces  $V_b$  in the model and hereby the shrink-and-swell phenomenon.

In practice the water/steam circuit is closed and the steam flow is governed by several valves combined with pipe resistance. Therefore a variable  $k(t)$  expressing pipe conductance and valve strokes is introduced.  $\dot{m}_s$  is then given as:

$$\dot{m}_s(t) = k(t) \sqrt{p_s(t) - p_{dws}} \quad (10)$$

where the downstream pressure,  $p_{dws}$ , is the pressure in the feed water tank which is open and hence has ambient pressure,

$p_{dws} = p_a$ .  $p_s(t) - p_{dws}$  is the differential pressure over the steam supply line.

The final second order model has the form:

$$\dot{x} = f(x, u, d) \quad (11a)$$

$$y = c(x, u, d) \quad (11b)$$

where  $y = [p_s, L_w]^T$ ,  $x = [p_s, V_w]^T$ ,  $u = [\dot{m}_{fu}, \dot{m}_{fw}]$  and  $d = k$ . As the temperature of the feed water is controlled it can be assumed constant and therefore not included in  $d$ .

The linearised version of this model is:

$$\dot{x}(t) = Ax(t) + Bu(t) + B_d d(t) \quad (12a)$$

$$y(t) = Cx(t) + Du(t) + D_d d(t) \quad (12b)$$

where we note the direct term from the disturbance to the water level output, due to the shrink-and-swell phenomenon. The matrices are given as:

$$A = \begin{bmatrix} \frac{-\dot{m}_s^0}{\varrho_w^0 V_w^0 \left\{ \frac{dh_w}{dp_s} \right\}^0} \left( \frac{h_c^0 + 2(p_s^0 - p_a) \left\{ \frac{dh_s}{dp_s} \right\}^0}{2(p_s^0 - p_a)} \right) & 0 \\ -\dot{m}_s^0 \left( 1 - \frac{h_c^0 + 2(p_s^0 - p_a) \left\{ \frac{dh_s}{dp_s} \right\}^0 \left\{ \frac{d\rho_w}{dp_s} \right\}^0}{\varrho_w \left\{ \frac{dh_w}{dp_s} \right\}^0} \right) & 0 \end{bmatrix} \quad (13a)$$

$$B = \begin{bmatrix} \frac{\eta}{\varrho_w^0 V_w^0 \left\{ \frac{dh_w}{dp_s} \right\}^0} & \frac{-(h_w^0 - h_{fw})}{\varrho_w^0 V_w^0 \left\{ \frac{dh_w}{dp_s} \right\}^0} \\ -\eta \left\{ \frac{d\rho_w}{dp_s} \right\}^0 & \left( 1 + \frac{(h_w^0 - h_{fw}) \left\{ \frac{d\rho_w}{dp_s} \right\}^0}{\varrho_w^0 \left\{ \frac{dh_w}{dp_s} \right\}^0} \right) \end{bmatrix} \quad (13b)$$

$$B_d = \begin{bmatrix} \frac{-h_c^0 \sqrt{p_s^0 - p_a}}{\varrho_w^0 V_w^0 \left\{ \frac{dh_w}{dp_s} \right\}^0} \\ \frac{-\sqrt{p_s^0 - p_a}}{\varrho_w^0 - \varrho_s^0} \left( 1 - \frac{h_c^0 \left\{ \frac{d\rho_w}{dp_s} \right\}^0}{\varrho_w^0 \left\{ \frac{dh_w}{dp_s} \right\}^0} \right) \end{bmatrix} \quad (13c)$$

$$C = \begin{bmatrix} 1 & 0 \\ \frac{\dot{m}_s^0 T_d}{A_{ws} \{\varrho_s^0\}^2} \left( \frac{\varrho_s^0 - 2(p_s^0 - p_a) \left\{ \frac{d\rho_s}{dp_s} \right\}^0}{2(p_s^0 - p_a)} \right) & \frac{1}{A_{ws}} \end{bmatrix} \quad (13d)$$

$$D = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix}, \quad D_d = \begin{bmatrix} 0 \\ \frac{T_d \sqrt{p_s^0 - p_a}}{A_{ws} \varrho_s^0} \end{bmatrix} \quad (13e)$$

where we have used  $\dot{m}_{fw}^0 = \dot{m}_s^0$ . Remember also that  $V_w^0 = A_{ws} L_w^0 + V_o - \frac{T_d}{\varrho_s^0} \dot{m}_s^0$ . We see that the linear model matrices depend

only on the pressure, the water level and the steam load. In particular, we see that the matrices are linearly dependent on  $\dot{m}_s^0$  if variations in  $V_w^0$  can be ignored. For reference the Laplace transform of the model is:

$$y(s) = G(s)u(s) + G_d(s)d(s) \quad (14)$$

with  $G(s) = C(sI - A)^{-1}B$  and  $G_d(s) = C(sI - A)^{-1}B_d + D_d$ . The complexity in this model is introduced because we insist on modelling the cross terms as well. However, a very good approximation using only direct terms can be given as:

$$p_s(s) = \hat{G}_{11}(s)\dot{m}_{fu}(s) + \hat{G}_{d,11}(s)k(s) \quad (15a)$$

$$L_w(s) = \hat{G}_{22}(s)\dot{m}_{fw}(s) + \hat{G}_{d,21}(s)k(s) \quad (15b)$$

with

$$\hat{G}_{11}(s) = \left( \frac{\eta}{\varrho_w^0 V_w^0 \left\{ \frac{dh_w}{dp_s} \right\}^0} \right) \frac{1}{s} \quad (16a)$$

$$\hat{G}_{22}(s) = \left( \frac{1}{A_{ws}(\varrho_w^0 - \varrho_s^0)} \right) \frac{1}{s} \quad (16b)$$

$$\hat{G}_{d,11}(s) = \left( \frac{-h_c^0 \sqrt{p_s^0 - p_a}}{\varrho_w^0 V_w^0 \left\{ \frac{dh_w}{dp_s} \right\}^0} \right) \frac{1}{s} \quad (16c)$$

$$\hat{G}_{d,21}(s) = \left( \frac{T_d \sqrt{p_s^0 - p_a}}{A_{ws} \varrho_s^0} \right) \frac{s - \frac{\varrho_s}{T_d(\varrho_w - \varrho_s)}}{s} \quad (16d)$$

The only unknown parameters in the presented model are the efficiency  $\eta$  and the residence time of the steam bubbles in the drum  $T_d$ . Normally when finding these one has a good idea about the size of the efficiency whereas regarding  $T_d$  one must rely on experience from previous boiler designs.

### 3 LIMITS OF PERFORMANCE

This section discusses in detail the properties of the marine boiler system and what sets the limit of how good performance can be achieved.

#### 3.1 Boiler nonlinearities

In [1] the MISSION<sup>TM</sup> OB boiler was shown to behave linearly over a large frequency range when varying the steam load. The nonlinearities present were mainly pronounced at low frequencies. This was seen as higher gains at lower steam load and variation of certain dynamics. The model used was of eighth order. The variation closest to the desired crossover frequency is that induced by the energy balance. This is also captured by the derived second order model. In particular, we refer to the pole presented in (13a) entry  $A_{11}$ . This corresponds to dynamics with a time constant that can vary between  $\approx 1000$  s (maximum load) to  $\approx 3500$  s (minimum load). Also the right half plane zero from fuel to water level varies in this frequency range from  $s \approx 0.0006$  (low load) to  $s \approx 0.003$  at high load. Also above these frequencies the gains both from inputs and disturbance to the outputs in the different load situation are coinciding. These properties are easily identified from the Bode plot of the derived second order model presented in Figure 2 for three different steam loads. We note that these variations of gain and dynamics are not present in the coupling from feed water to water level.

Regarding the singular values the linear range is at angular frequencies  $> 0.003 \frac{\text{rad}}{\text{s}}$  which includes the desired crossover also one linear controller has proved to behave similar over the entire load range. For these reasons it was concluded in [1] that a controller design could rely on one linear model if the controller had integral action. If attention is directed at the phase of the different transfer functions in the model this range starts at a higher frequency  $\approx 0.01 \frac{\text{rad}}{\text{s}}$  except for the transfer function from fuel to water level where phase differences over a large frequency range still persist, see Figure 2. This means that when designing diagonal controllers the nonlinearities of the plant pose no limit on the performance. When designing multiple input multiple output controllers one must keep in mind that the phase response of the transfer function from fuel to water level is inaccurate when relying on one lin-

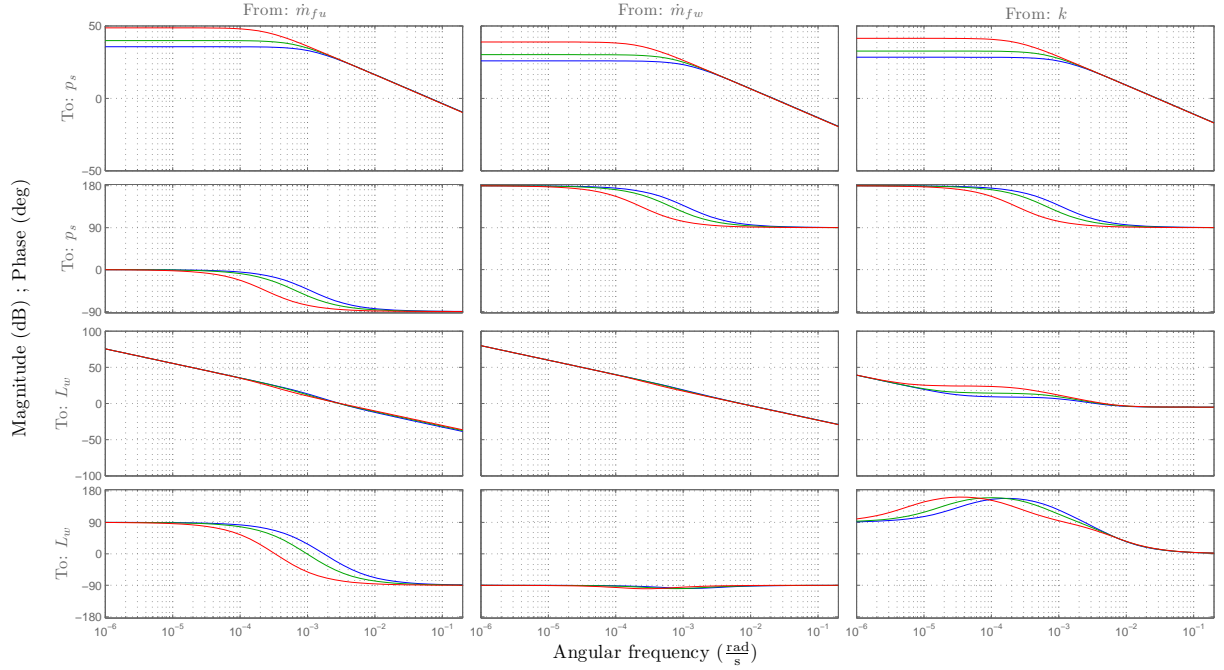


Figure 2: Bode plot of scaled  $[G(s) G_d(s)]$  for three different steam loads. Blue:  $\dot{m}_s = 1800 \frac{kg}{h}$ , green:  $\dot{m}_s = 1100 \frac{kg}{h}$  and red:  $\dot{m}_s = 400 \frac{kg}{h}$ . Note the variation of gain at low frequencies to the pressure output and the change in system bandwidth.

ear model for design.

In [5] it was pointed out that in certain situations, e.g. when using hysteresis control where the system state never reaches a steady state but rather converges to a limit cycle, then the low frequency nonlinearities can cause problems. First of all the limit cycle which the state converges to will be dependent on the low frequency parts of the model if the switching does not occur too frequent, and further an estimate of the steam flow is difficult to derive and is very slowly converging when using only one linear model.

This indicates that dependent on the controller strategy, the nonlinearities are more or less important. However, hysteresis control is only applied in very special cases, and normally the low frequency gain variations can be ignored when designing controllers.

In the above we have suggested that the boiler system is described sufficiently using a linear model. However, this model does not include the actuator systems generating the fuel and feedwater flows. These actuator systems are nonlinear and will be considered next.

### 3.2 Actuator systems

#### 3.2.1 Feed water supply system

The feed water system is well known and a diagram of it is shown in Figure 3.

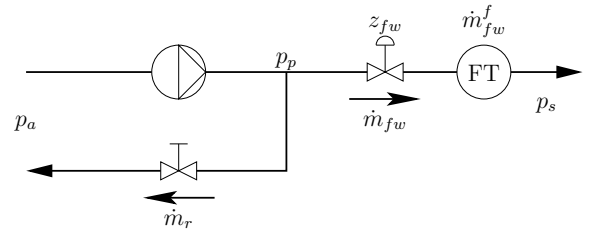


Figure 3: Diagram of feed water system. Water pumped from the feed water tank is injected into the boiler in the forward path, and in the return path the water is led back to the feed water tank.

The valve in the forward path is a pneumatic control globe valve which has an equal percentage characteristic (chosen over the linear characteristics as it in this setup helps linearising the gain from stroke to flow). The flow through the valve can be expressed as [22]:

$$\dot{m}_{fw} = k_f f(z_{fw}) \sqrt{p_p - p_s} \quad (17)$$

where

$$f(z_{fw}) = \frac{1}{R}(R^{z_{fw}} - e^{-R_0 z_{fw}}) \quad (18)$$

is the function describing the valve characteristic and relating the valve stroke,  $z_{fw}$ , to the flow.  $k_f$  is the valve gain, which in the valve data sheet is usually given for water at 20 °C and expresses the flow through a fully open valve with a pressure drop of 1 bar over the valve. Finally  $p_p$  is the pressure after the pump and  $p_s$  is the steam pressure in the boiler. In most cases the dynamics that govern the feed water supply system is that of the flow sensor as the pneumatic control valve and flow dynamics are fast when pipes are not too long. The sensor may adequately be described with dynamics of first order given as:

$$\dot{m}_{fw}^f(s) = G_{fw}(s) \dot{m}_{fw}(s) = \frac{1}{\tau_{fw}s + 1} \dot{m}_{fw}(s) \quad (19)$$

The valve in the return path is a manually adjustable valve which should not be adjusted during plant operation. The flow through this valve is expressed as:

$$\dot{m}_r = k_r \sqrt{p_p - p_{fw}} \quad (20)$$

where  $k_r$  is the return valve conductance and  $p_{fw}$  is the pressure in the feed water tank assumed equal to the ambient pressure  $p_{fw} = p_a$ . The function of this valve is to change the characteristics of the pump which is running at fixed speed. The flow delivered by the pump is:

$$\dot{m}_p = \dot{m}_r + \dot{m}_{fw} \quad (21)$$

and the pressure after the pump and before both the forward and the return valves is:

$$p_p = \Delta p_p + p_a \quad (22)$$

where  $\Delta p_p$  is the pressure rise over the pump which again can be found as:

$$\Delta p_p = \rho g \Delta H_p \quad (23)$$

where  $H_p$  is the lifting height. This height is often approximated in the literature, see

e.g. [23, p.], by:

$$\Delta H_p = H_{p,max} \left( \left( \frac{n_p}{n_{p,max}} \right)^2 - \left( \frac{Q_p}{Q_{p,max}} \right)^2 \right) \quad (24)$$

where  $H_{p,max}$  is the maximum lift height occurring at a zero throughput.  $n_p$  and  $n_{p,max}$  is the current and maximal pump speed, respectively, and  $Q_p$  and  $Q_{p,max}$  is the current and maximal flows. Note that  $Q_p = A v_p$  and  $\dot{m}_p = \rho_w A v_p$ , where  $A$  is effective flow area and  $v_p$  is the pump speed. This together with (23) leads to:

$$\Delta p_p = p_{p,max} \left( \left( \frac{n_p}{n_{p,max}} \right)^2 - \left( \frac{\dot{m}_p}{\dot{m}_{p,max}} \right)^2 \right) \quad (25)$$

The pump always runs at maximum speed meaning that we can simplify the expression by using:  $n_p = n_{p,max} \Leftrightarrow n_p/n_{p,max} = 1$ .

We want a model of the feed water supply system that gives us a feed water flow when we send a certain voltage to the control valve. The equations above, however, do not allow to put up such a model in a straight forward manner, (insertion of (17) and (20) into (25) and isolating for  $\dot{m}_{fw}$  requires solving a quadratic equation), this is treated next.

### 3.2.2 Explicit expression for feed water flow

First we notice that the feed water flow to the boiler,  $\dot{m}_{fw}$ , is dependent on three parameters; the control variable,  $z_{fw}$  (the valve stroke), the steam pressure,  $p_s$ , which acts as a disturbance and finally the stroke position of the manual valve in the return path which together with the valve characteristics is described by  $k_r$ .

The explicit expression for the feed water flow as function of the valve stroke and boiler pressure is now found as

$$\dot{m}_{fw} = g(z_{fw}, p_s) = k_f f(z_{fw}) \sqrt{\Delta p_p + p_a - p_s} \quad (26)$$

where  $\Delta p_p$  is given as the solution to a quadratic equation:  $\Delta p_p = \frac{-a_1 - \sqrt{a_1^2 - 4a_2a_0}}{2a_2}$



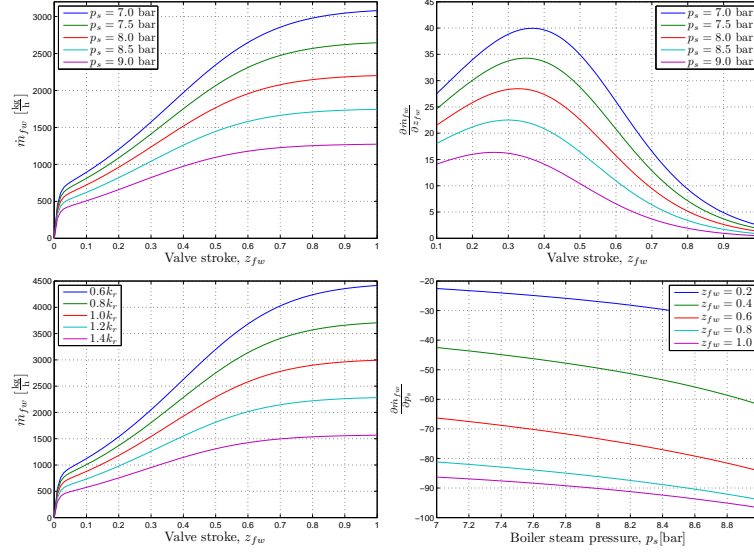


Figure 4: Feed water system characteristics. In the top left corner the feed water flow is shown as a function of the valve stroke for different boiler pressures. In the top right corner the partial derivative of the feed water flow with respect to the valve stroke is shown for different boiler pressures. Notice that only valve strokes  $z_{fw} \geq 0.1$  are included as the valve positioning is unreliable below this level. In the bottom left corner the feed water flow is shown as a function of the valve stroke for different return valve strokes, and in the bottom right corner the partial derivative of the feed water flow with respect to the boiler pressure is shown for different valve strokes.

with:

$$a_2 = \left\{ \begin{aligned} & \left[ 1 + \frac{p_{p,max}}{\dot{m}_{p,max}^2} (k_r^2 + k_f^2 f(z_{fw})^2) \right]^2 + \\ & -4 \frac{p_{p,max}}{\dot{m}_{p,max}^4} k_r^2 k_f^2 f(z_{fw})^2 \end{aligned} \right\} \quad (27a)$$

$$a_1 = \left\{ \begin{aligned} & 2 \left[ 1 + \frac{p_{p,max}}{\dot{m}_{p,max}^2} (k_r^2 + k_f^2 f(z_{fw})^2) \right] \times \\ & \left( \frac{p_{p,max}}{\dot{m}_{p,max}^2} k_f^2 f(z_{fw})^2 (p_a - p_s) - p_{p,max} \right) + \\ & -4 \frac{p_{p,max}}{\dot{m}_{p,max}^4} k_r^2 k_f^2 f(z_{fw})^2 (p_a - p_s) \end{aligned} \right\} \quad (27b)$$

$$a_0 = \left( \frac{p_{p,max}}{\dot{m}_{p,max}^2} k_f^2 f(z_{fw})^2 (p_a - p_s) - p_{p,max} \right)^2 \quad (27c)$$

This model provides a good fit to measurement data and the sensor time constant is about  $\tau_{fw} = 4$  s modelled in by the first order system  $G_{fw}(s)$  (19). Note that the pipe resistance from the valve to the boiler has been ignored. The only unknown parameter in this model is the positioning of the return valve  $k_r$ . It is obvious that the system is very nonlinear which is illustrated in Figure 4.

From the figure it can be seen that for the example boiler pressure the small gain

(top right plot) can vary worst case up to a factor of 35 and at nominal pressure 8 bar up to a factor of 22.

Unfortunately, the flow is dependent on the boiler steam pressure. One implication from this is that the upper achievable flow is dependent on this disturbance  $\dot{m}_{fw} \in [0, \bar{m}_{fw}(p_s)]$ . This means that if one uses flow as control variable in an outer loop together with a flow controller the input constraints are not constant which might cause problems in e.g. model predictive controller (MPC) configurations.

As always there are two possibilities regarding control. Either try to linearise the actuator dynamics by flow feedback or gain scheduling or use the valve stroke directly as a control variable in the outer loop. Using the valve stroke directly in the outer loop has two major disadvantages.

First of all the disturbance from varying boiler steam pressure is not compensated for. Worst case this can result in an unintentional coupling as e.g. an increase in pressure causes the feed water flow to decrease. This will cause a level controller

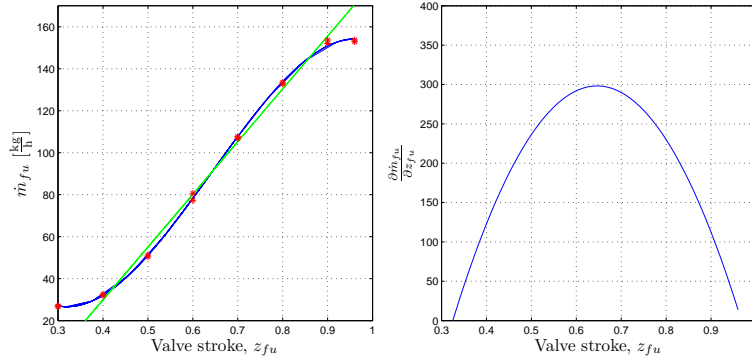


Figure 5: Fuel system characteristic. The red dots are measurement points, the blue curve is a third order polynomial fitted to the measurements and the green line is a first order fit to the data set truncated to valve strokes  $z_{fu} \in [0.4, 0.9]$ .

to open the feed water valve more which will increase the feed water inlet but at the same time the extra water causes the pressure to drop and hence the feed water flow to increase even more. This phenomenon is especially pronounced if the gap between level and pressure loop bandwidth is small.

The other disadvantage is that when using such a strategy it is custom to design the feedback controller according to the largest gain (see Figure 4 top right). However, with such large gain variations this means that the actually achieved bandwidth may vary more than one decade. Here we have assumed a slope  $20 - 40 \frac{\text{dB}}{\text{dec}}$  around the crossover frequency, which is consistent with what would be achieved by PI control on (16b). If a flow sensor is available this is an unnecessary restriction as the valve gain can be linearised by feedback or gain scheduling. Using only feedback the bandwidth of the flow dynamics will vary just as much as those for the outer controller in the above example. However, this might be acceptable as this inner loop can be made very fast, meaning that if the outer level loop bandwidth is not too high the effort of designing a gain scheduling to linearise the flow dynamics is not worthwhile. Pure gain scheduling and no feedback is not preferable as this still leaves the problem of compensating for the boiler pressure disturbance and it requires a very accurate model of the system gain.

The feed water system is designed in

such a way that the maximal flow is higher than the maximal steam production at nominal pressure. This insures that instability of the water level is avoided due to constraint limitations in the flow.

### 3.2.3 Burner

The burner must deliver the requested power while keeping a clean combustion. The dynamics of combustion is very fast and we saw earlier that the heat released in the combustion is treated stationary and proportionally to the fuel flow (1). The burner systems is not as well known as the feed water system and it has not been possible to acquire information on the functionality of the nozzle-lance/atomiser system. For this reason no first principle model has been derived of this unit. However, data-sheets of the atomiser and measurements suggest a third order characteristic between valve position,  $z_{fu}$ , and flow  $\dot{m}_{fu}$ . In Figure 5 left a third order fit, blue, to measurement data, the red dots, is shown together with a linear fit to the region  $z_{fu} \in [0.40, 0.90]$ .

In the right plot of the figure the derivative of the flow with respect to the valve stroke is shown. From the plot it can be seen that in the region  $z_{fu} \in [0.4, 0.9]$  the gain varies up to a factor of 2.5. Designing a robust controller with reasonable stability margins such a gain variation is not a problem. Small boilers like the one treated in this paper are never shipped with fuel flow sensors as this is too expensive. For this reason no feedback can be

closed around the fuel system to linearise the gain. However, as can be seen from Figure 5 the linear approximation is good in a large operating range and represents a gain not far from the maximal. It is assumed that such a relationship can be found by simple experiments. This leaves the fuel system controller as a pure feed-forward control of the flow, and slow controller response has to be accepted when in the low flow range below a valve stroke of 0.4 and in the high flow range above a valve stroke of 0.9. The valve position is adjusted by an electric motor using pulse width modulation (PWM) of the electric control signals. The behaviour of this controller is dead beat.

Keeping a clean combustion is a matter of having the correct fuel to air ratio. In boilers treated here the combustion air flow is controlled by letting the air damper position be directly dependent on the fuel control valve position.

The dynamics between fuel flow and position are negligible. However, the electric motor controlling the position only has one speed which sets a rate constraint on the change in fuel flow.

### 3.2.4 Input constraints

Both the feed water and fuel system are subject to constraints and these constraints are likely to be active during disturbance rejection and reference tracking. These constraints set the limit for how fast the disturbances can be rejected or how fast the setpoint can be changed. The constraints are never active during normal steady state operation unless the steam flow is so low that on/off burner control is necessary.

Regarding reference changes the limit on the rate of change depends on the steam load, e.g. if close to the maximum steam consumption there will not be much excess feed water or fuel to fast increase the water level or pressure.

Regarding disturbance rejection the same holds. However, here the nonlinearities introduced by the constraints will become important. This is due to the non-minimum phase zero in the response from steam flow to water level, shrink-and-

swell. These issues are discussed in the following section.

Due to the frequent activation of constraints during disturbance and reference changes and the need for integral action to avoid steady state offsets it is important to include an appropriate anti-windup scheme in the controller design.

### 3.3 Shrink-and-swell

The worst shrink-and-swell behaviour is a consequence of changes in the steam load disturbance. As illustrated in Section 2 the bandwidth of this disturbance is very high and we neglected the dynamics of the bubble volume. To fully cancel the transients in the water level during step changes in the disturbance input one can look at the change in volume of the water and steam bubble mixture. Then for any disturbance we need  $Q \geq \frac{d(V_w + V_b)}{dt}$  where  $Q$  is a volume flow added to the process. When a steam flow disturbance occurs the short term changes are spotted in the steam bubble volume according to (9). For instance a step in steam flow of  $\Delta\dot{m}_s$  causes a change in bubble volume of  $\Delta V_b \approx 0.63 \frac{T_d}{\varrho_s} \Delta\dot{m}_s$  in  $T_d$  seconds. This would require  $Q \geq 0.63 \frac{1}{\varrho_s} \Delta\dot{m}_s$  which depends only on steam data and hence the pressure and the size of the step in steam flow. A small step in steam flow of  $\Delta\dot{m}_s = 100 \frac{\text{kg}}{\text{h}}$  requires  $Q$  to be higher than  $15 \frac{\text{m}^3}{\text{h}}$  with a very high rate of change. It is of course not possible to generate such flow rates to or from the boiler. Further we did not specify that this flow must be water or steam at saturated conditions as the response from feed water to water level has a right half plane zero at high frequencies. Also attempts to generate such flow rate would compromise pressure performance.

This means that one has to accept the shrink or swell as a consequence of disturbance changes. However, the rate of recovery from these can be tuned. Our model (9), (16d) gives us a rough estimate of the shrink-and-swell occurring as  $\Delta L_w = \frac{T_d}{A_{ws} \varrho_s} \Delta\dot{m}_s$  if we assume that the reference is reached between steps in the disturbance. For the worst case step from 400-1800  $\frac{\text{kg}}{\text{h}}$  of steam this corresponds to a

swell of 5.6 cm.

As mentioned we cannot cancel the effect of the steam flow disturbance completely. Even so any linear feedback regulator will try. E.g. for a large positive disturbance step the swell will cause the controller to lower the feed water input. Often it will reach the lower zero flow constraint. However, the swell is followed by the negative integrating response from the disturbance. This response is fast making it difficult for the controller to avoid undershoot in the response. To avoid such problems a feedforward or fast estimate of the steam flow is necessary.

As mentioned above there is a right half plane zero in the response from feed water to water level which is also a consequence of shrink-and-swell. This zero was not included in the model in Section 2. The reason is that higher order models place this zero at a frequency much higher than the desired crossover frequency. Further this zero has been difficult to spot in measurement performed on the full-scale boiler.

### 3.4 Disturbances

The most important disturbance to the pressure and level is the steam flow which was treated above. Other disturbances as feed water temperature, fuel temperature, combustion air temperature and ambient pressure do not affect these outputs much. This was shown in [1] for the feed water and fuel temperatures. These two are also controlled in the plant. All these disturbances enter at the plant input and can be seen as unmeasured disturbances in the firing rate/fuel flow.

### 3.5 Neglected dynamics

As mentioned in Section 2 much of the plant high frequency dynamics have been neglected and also some of the low frequency dynamics have been neglected in the second order control model. The low frequency dynamics came from the dependency of the efficiency on the water level, which we already pointed out only moved the integrator for the water level slightly into the left half plane and added a zero in the origin from fuel flow to pressure. Regarding the zero in the origin for the pressure this is removed when the level is

controlled and the pole moved into the left half plane is still close to the origin so that the dynamics behave as an integrator close to the crossover frequency.

The high frequency dynamics were actuator dynamics, flue gas dynamics, metal dynamics and bubble volume dynamics. Obviously these could be included in the control model reducing uncertainty, however, at the expense of higher model and controller order when using a model-based approach. If the desired crossover frequency is close to the bandwidth of the neglected dynamics the controller must exhibit appropriate stability margins or the model expanded. However, as long as the bandwidth is kept below  $0.1 \frac{\text{rad}}{\text{s}}$  and a reasonable stability margin is attained the models (12) or (15) can be used for controller design.

### 3.6 Decentralised control and interaction

In [1] it was shown that the interaction in the system does not cause any stability problems for a diagonally designed controller. Also it was shown that benefits especially in pressure performance could be expected by applying a multiple input multiple output (MIMO) control to the process. However, in practise these benefits were shown in [3] not to be the main advantage of MIMO control. It was shown that due to noise on the water level measurement the bandwidth of the response from the steam flow disturbance to the water level using SISO control was limited. However, using a model-based MIMO controller improvements were shown. This was seen as an improved steam flow disturbance rejection on the water level compared with SISO PI controllers.

### 3.7 Uncertainty

In [3] it was mentioned that test had exposed unexpected uncertainty in the cross-couplings of the presented model. This led to a poor controller performance when the feed water actuator was assigned a low weight in the performance index of an optimising controller. However, simulations have shown that this is less severe as increasing the weight on the feed water flow did not make any visible deterioration compared to the desired performance and

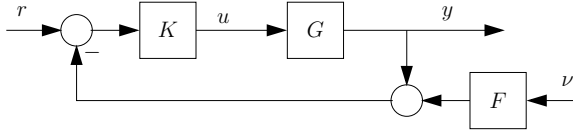


Figure 6: Block diagram of closed loop system with measurement noise input filter.

at the same time increased robustness.

### 3.8 Measurement noise

The measurement noise in the system is what really set the limit of the achievable performance. Especially the water level, which is measured by a device using a capacitive measurement principle, is subject to much noise caused by turbulence in the water surface. Analysing measurement data collected from the full-scale boiler shows that the measurement noise for both pressure and water level can be modelled as unit variance zero-mean white Gaussian low pass filtered noise processes where the actual noise variances have been moved into the filters. Such filters are found by identifying autoregressive models for each noise channel. For the steam pressure this results in the first order filter  $F_{ps}(s)$  and for the water level the second order filter  $F_{Lw}(s)$ . The noise input is  $\nu = [\nu_1, \nu_2]^T$  with  $E\{\nu(t)\} = 0$ ,  $E\{\nu(t_1)\nu^T(t_2)\} = \delta(t_2 - t_1)I$ . See Figure 6 for reference on where the noise enters the system.

Figure 7 illustrates the problems introduced by the noise. The figure shows the control sensitivity function  $M(s) = K(s)(I + G(s)K(s))^{-1}$  achieved by the diagonal PI controller  $K(s)$  for two different settings of proportional and integral terms. The controllers were designed from (16) to achieve a phase margin of  $71^\circ$  and certain crossover frequencies. Particularly, the crossover frequency for the pressure loop is in both designs is  $\omega_{c,ps} = 0.075 \frac{\text{rad}}{\text{s}}$  whereas it for the level loop in design  $K_1(s)$  was  $\omega_{c1,Lw} = 0.0068 \frac{\text{rad}}{\text{s}}$  and in design  $K_2(s)$  was  $\omega_{c2,Lw} = 0.021 \frac{\text{rad}}{\text{s}}$ .

The figure further displays the noise filters  $F(s) = \text{diag}(F_{ps}(s), F_{Lw}(s))$  and product of the filters and control sensitivity  $M(s)F(s)$ . It is important to remem-

ber that neither of the displayed transfer functions are scaling independent for which reason appropriate scaling of input and noise must be applied. The noise was already scaled and the input is scaled according to allowed input variation.

From the figure it can be seen that the noise on the water level causes problems for the controller. It can also be seen that increasing the crossover frequency of the level controller from  $\omega_{c1,Lw} = 0.0068 \frac{\text{rad}}{\text{s}}$  the measurement noise will cause large control signal action.

This is a problem as the bandwidth of the disturbance response is high and as a result we get a slow regulation and long settling between disturbances changes.

In fact it is very difficult to push the bandwidth of the level loop above  $\omega_{c,Lw} = 0.0068 \frac{\text{rad}}{\text{s}}$  and still keep a reasonable control signal. However, by careful design of measurement filters small improvements can be achieved. In particular, an LQG design has shown capable of achieving a  $40 \frac{\text{dB}}{\text{dec}}$  slope just above the chosen crossover frequency and reasonable control signals can be achieved with the crossover frequency at  $\omega_{c,Lw} = 0.01 \frac{\text{rad}}{\text{s}}$ . This can be done by keeping an appropriate stability margin without adding extra model states apart from those introduced to achieve integral action. Similar performance can be achieved by designing a second order filter in combination with a PI design. However, pushing the crossover frequency has a negative effect on the pressure performance when using a diagonal controller, and further this controller will be of approximately the same order as an LQG compensator.

### 3.9 Output constraints

Hard constraints are present on both the water level and pressure for the marine boiler. The high pressure constraint is important but not likely to become active unless a fault has occurred in the system. Regarding the water level both upper and lower alarms can be present on the boiler. There is a demand from the classification societies that the flue gas pipes must be under water up to the point at which the flue gas drops below  $600^\circ\text{C}$ .

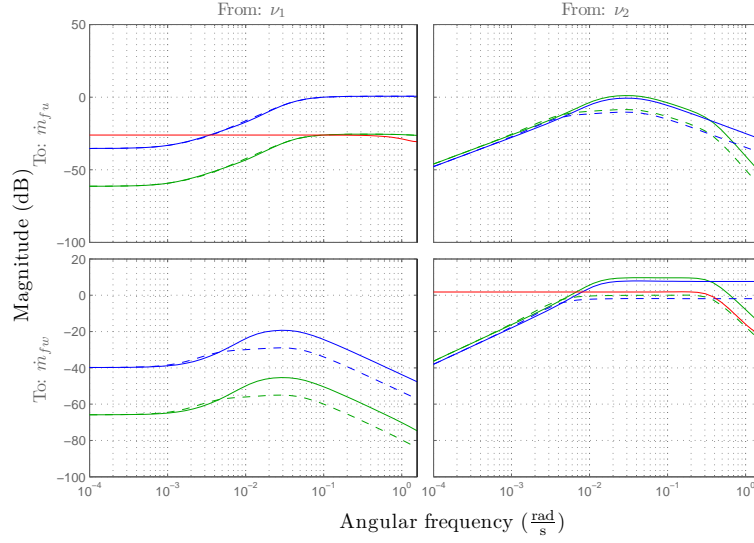


Figure 7: Magnitude plot of noise shaping filters  $F(s)$ , red, control sensitivity function  $M(s)$ , blue, and product between control sensitivity function and shaping filters  $M(s)F(s)$ , green. The dashed lines corresponds to design  $K_1(s)$  and the solid lines design  $K_2(s)$ . Notice the high gains in the bottom right plot from  $\omega \approx 0.007$  to  $\omega \approx 0.7$ .

This sets requirements for the fluctuation on the water level in the drum. Of course, this bound must in any case be somewhat conservative as it is at a constant level whereas the point would in reality change with load. The boiler is equipped with a low water level alarm to indicate when the water level is within a certain range from this point (typically 45-60 mm). In some cases the boiler is also equipped with a high water level alarm to prevent water from entering the steam supply line and keep a good steam quality. This is especially important when the downstream equipment is turbines.

These constraints must not become active for which reason the distances to the high and low alarms from normal water level operation are designed somewhat conservative today. In the introduction we mentioned that there is a desire to reduce the physical geometry of the boiler. This can be achieved by reducing the distances between the high and low water level alarms. We postulate that this cannot be achieved by feedback alone but must be accompanied by a water level setpoint controller. The reason is the shrink-and-swell phenomenon and the fact that the steam flow disturbance is not known

in advance. As was illustrated in Section 3.3 the level variation caused by shrink-and-swell has to be accepted. But by appropriate feedback (and possible feed-forward) the recovery time from a step can be reduced and especially the overshoot/undershoot when rejecting the disturbance can be eliminated such that the maximum variation is not increased if a step in the disturbance is applied in the opposite direction before the level has settled again. By augmenting the feedback structure by a setpoint algorithm maximising the level at all time by estimating the current worst case disturbance there is a possibility to reduce the distance between the high and low water level alarms. However, remember that the worst shrink or swell caused a level variation of approximately 5.6 cm (Section 3.3) meaning that under perfect control the potential maximal level variation could be reduced to 5.6 cm. More precise we can write water level constraints equations based on the current worst case disturbance. Define HWL as the water level at which the high water level alarm is activated and LWL as the water level at which the low water level alarm is activated. Then the maximum allowable water level,  $L_{w,max}$ , at any instant

is given by:

$$\begin{aligned}
L_{w,max} &= HWL - \frac{\Delta V_{b,max}(\dot{m}_s)}{A_{ws}} \\
&= HWL - \frac{T_d}{A_{ws}\rho_s} \Delta \dot{m}_{s,max}(\dot{m}_s) \\
&= HWL - \frac{T_d}{A_{ws}\rho_s} (\dot{m}_{s,max}(\dot{m}_s) - \dot{m}_s) \\
&= HWL - \frac{1}{A_{ws}} (V_{b,max}(p_s) - V_b)
\end{aligned} \tag{28}$$

where  $\Delta V_{b,max}(\dot{m}_s)$  denotes the maximum positive change in bubble volume given the current disturbance. Likewise  $\Delta \dot{m}_s$  denotes the maximum possible positive change in disturbance at current time. Also  $V_{b,max}(p_s)$  denotes the maximum volume at of steam below the water surface given the maximum disturbance  $\dot{m}_{s,max}$ .  $V_b$  here denotes the current bubble volume. This relationship is only possible due to linearity of the steam bubble volume in the load assumed in (9). This leads to the following water volume constraint:

$$\begin{aligned}
L_w &\leq HWL - \frac{1}{A_{ws}} (V_{b,max}(p_s) - V_b) \\
\frac{V_w - V_b}{A_{ws}} &\leq HWL - \frac{1}{A_{ws}} (V_{b,max}(p_s) - V_b) \\
V_w &\leq A_{ws}HWL - V_{b,max}(p_s)
\end{aligned} \tag{29}$$

Likewise for the low water level constraint we get:

$$L_{w,min} = LWL + \frac{1}{A_{ws}} (V_b - V_{b,min}(p_s)) \tag{30}$$

leading to the water volume constraint:

$$V_w \geq A_{ws}LWL - V_{b,min}(p_s) \tag{31}$$

more compact this gives the water volume constraint

$$\left( \frac{A_{ws}LWL - V_{b,min}(p_s)}{A_{ws}} \right) \leq V_w \leq \left( \frac{A_{ws}HWL - V_{b,max}(p_s)}{A_{ws}} \right) \tag{32}$$

For perfect control of the water volume the minimum distance between the HWL and

LWL is then:

$$\begin{aligned}
HWL - LWL &\geq \frac{(V_{b,max}(p_s) - V_{b,min}(p_s))}{A_{ws}} \\
&\geq \frac{\Delta V_{b,max}(\dot{m}_s)}{A_{ws}} = \frac{T_d}{A_{ws}\rho_s} \Delta \dot{m}_{s,max}(\dot{m}_s)
\end{aligned} \tag{33}$$

This essentially means that we should control the water volume in the boiler and not the actual water level. In [19] the authors define a narrow range water level as the water level which includes the bubble volume and a wide range water level as one that only measures the water in the drum. A measurement of the wide range water level can be generated by a differential pressure measurement as suggested in [19, 21]. In [19] they end up controlling the narrow range water level which must be kept within pre-calculated alarm levels to ensure that the wide range water level is high enough. For the one-pass smoke tube boiler it seems more appropriate to control the wide range water level as this can be done without any fast feedback. However, due to model uncertainties it is still important to have constraints on the narrow range water level to ensure good steam quality and avoid violation of low water level constraints.

## 4 CONTROLLER DESIGN GUIDELINES

This section is devoted to present simple control design guidelines for the one-pass smoke tube boiler. The control scheme suggested uses a cascade configuration with actuator flow controllers in an inner loop and outer controllers handling pressure and level control.

There are two reasons for choosing such a configuration; first of all the feed water valve system is difficult to describe and highly nonlinear and closing the loop will partly linearise the map from feed water reference to actual flow, secondly this approach helps minimising uncertainties at the boiler plant input. However, the upper feed water flow bound is still dependent on the boiler pressure which might cause trouble if designing e.g. an MPC

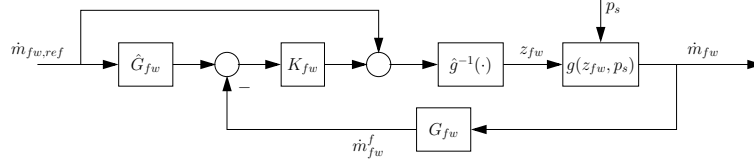


Figure 8: Feed water control scheme including both feedforward and feedback.  $K_{fw}$  is the feedback controller (a PI controller),  $\hat{g}^{-1}$  is a model of the feed water system gain and  $\hat{G}_{fw}$  is a model of the feed water sensor dynamics. Notice for  $\hat{G}_{fw} = G_{fw}$  and  $\hat{g} = g$  we have  $\dot{m}_{fw}^f = G_{fw}\dot{m}_{fw,ref}$ .

controller in which the constraints are assumed to be known.

#### 4.1 Actuators

The fuel flow controller is based on pure feedforward. As described in Section 3.2 this feedforward is based on a linear map which is a good approximation of the actual map from fuel flow reference to valve position.

Regarding the feed water it was partly illustrated in [20] that closing a loop around the feed water flow can limit the variance of the controlled water level and pressure. The reason for this observation is most likely the coupling from the boiler steam pressure to the feed water flow discussed in Section 3.2.

The general structure of the feed water controller we consider here is shown in Figure 8.

When using neither feedback or gain scheduling for flow control we have to design the controller somewhat conservative to be able to handle the large gain variations. This could maybe be excepted if it had not been for the large influence of the measurement noise which set an upper bound for the bandwidth. Already this bandwidth is very low meaning that the level loop bandwidth will become extremely low.

For these reasons it will be advantageous to include a feedback to linearise the gain and provide robustness against the pressure disturbance. Now as the measurement noise set a low achievable bandwidth feedback will in most cases provide adequate performance as the variations in time constant for the closed loop feed water system is such that the lowest time con-

stant still corresponds to dynamics faster than the level loop. However, if another level measurement becomes available, gain scheduling will become necessary to be able to raise the bandwidth. For this reason we shortly discuss the inverse of the map from feed water control valve to feed water flow.

Dividing the control system into different modules also makes the design more flexible as the outer controller becomes somewhat independent on the burner and feed water system configuration and the other way around. Hence breaking the system into modules allows changing modules without influencing the complete control system.

##### 4.1.1 Inverse mapping

For control purpose we are interested in finding the inverse mapping of  $g(z_{fw}, p_s)$  which is a function mapping a reference flow and a particular boiler pressure to a valve stroke,  $g^{-1} : \mathbb{R}^2 \mapsto \mathbb{R}$ .

From (17) we immediately get:

$$z_{fw,ref} = f^{-1} \left( \frac{\dot{m}_{fw,ref}}{k_f \sqrt{p_{p,ref} - p_s}} \right) \quad (34)$$

which gives:

$$z_{fw,ref} = \frac{\log \left( \frac{\dot{m}_{fw,ref}}{k_f \sqrt{\Delta p_{p,ref} + p_a - p_s}} \right)}{\log(R)} + 1 \quad (35)$$

Now we need to find  $\Delta p_{p,ref}$  as a function of  $\dot{m}_{fw,ref}$ . To do so we proceed with the



following version of (25):

$$\Delta p_{p,ref} = \quad (36)$$

$$p_{p,max} \left[ 1 - \left( \frac{k_r \sqrt{\Delta p_{p,ref}} + \dot{m}_{fw,ref}}{\dot{m}_{p,max}} \right)^2 \right]$$

then  $\Delta p_{p,ref}$  is the solution to a quadratic equation:  $\Delta p_{p,ref} = \frac{-b_1 - \sqrt{b_1^2 - 4b_2b_0}}{2b_2}$  where:

$$b_2 = \left[ \left( 1 + \frac{p_{p,max}}{\dot{m}_{p,max}^2} k_r^2 \right)^2 \right] \quad (37a)$$

$$b_1 = \left[ \begin{aligned} & 2 \left( 1 + \frac{p_{p,max}}{\dot{m}_{p,max}^2} k_r^2 \right) \times \\ & \left( \frac{p_{p,max}}{\dot{m}_{p,max}^2} \dot{m}_{fw,ref}^2 - p_{p,max} \right) + \\ & -4 \frac{p_{p,max}^2}{\dot{m}_{p,max}^4} k_r^2 \dot{m}_{fw,ref}^2 \end{aligned} \right] \quad (37b)$$

$$b_0 = \left( \frac{p_{p,max}}{\dot{m}_{p,max}^2} \dot{m}_{fw,ref}^2 - p_{p,max} \right)^2 \quad (37c)$$

In [20] the inverse (34) was approximated by the solution to a quadratic equation in  $\dot{m}_{fw,ref}$  which proved to give satisfactory results in practice. The pressure dependency was omitted but treated as an unmeasured disturbance handled by feedback.

## 4.2 Boiler

The performance specifications for the oil-fired one-pass smoke tube boiler are vague. The actual steam consumption pattern on the ships is unknown but known to vary dependent on the type of vessel. Regarding the water level there are no consumer requirements but as discussed in Section 3.9 there is a wish from AI to minimise the level fluctuations which means that fast damping of the disturbance is needed and no overshoot can be tolerated. Regarding the pressure the same holds that there are no consumer specified performance demands. However, setting high demands for the water level performance one has to expect that this will come at the expense of the pressure performance. This does not mean that the pressure is allowed to vary arbitrarily. When the boiler is bought for heating in various application the steam output from the boiler is expected to have a certain

temperature which is directly equivalent to boiler pressure as it operates under saturated conditions. Further large fluctuations in the pressure and here by boiler construction temperature cause stress in material and a reduced product lifetime. Despite this lack of knowledge AI assumes that steps in the disturbance can occur every tenth minute.

The model (16) serves as a good candidate for designing classical controllers as PI controllers. This is also a consequence of the weak nonlinear behaviour of the plant around the crossover frequency as was discussed in Section 3.1. From this model it is easy to derive analytic expressions for the proportional and integral terms of the PI controller specifying design parameters such as desired crossover frequency and phase margin. Especially the phase margin can be chosen to account for neglected actuator and measurement filter dynamics. It would also be possible to use single input single output (SISO) model predictive controllers which handle the input constraints in a natural way. In [24] it was shown that such controllers have approximately the same computational burden as classical PID controllers.

It is advisable to include measurement filter of at least second order with a bandwidth not much over the desired crossover frequency to limit the large influence of the noise and keep adequate control signals.

If it is possible to create an estimate of the steam flow this is strongly advisable as this can be used in a feedforward to the level control especially to avoid overshoot and speed up rejection of the steam flow disturbance. A Kalman filter was shown in [2] to be able to generate such an estimate. But simpler estimates can be generated by considering the much faster pressure loop. The fuel flow must to some degree together with the current feed water flow give an estimate of the current steam flow (e.g. by considering a steady state version of (7)).

## 4.3 Controller tuning

It is of interest to reduce installation time of new boilers by making the control system auto tuning. This will further limit the time spent by service personal during

commissioning. Further this makes it possible to make the performance invariant to the environment into which the boiler is placed. During the boiler lifetime it is also likely that sensors or actuators are replaced, adjustment can be made to e.g. the return valve position  $k_r$  of the feed water system and different films might build up on both the water and flue gas sides of the heating surface. These things can change the boiler dynamics, and to keep appropriate controller performance a re-tuning might be necessary.

In [20] the first attempts to make the marine boiler control system tune automatically was made. In fact such a tuning can be made by just identifying a few model parameters. The reason here being the simple structure of  $\hat{G}_{11}$  and  $\hat{G}_{22}$  in (16) suggested to be used for controller design. There are no unknowns in  $\hat{G}_{22}$  which depend on construction data and operating conditions alone whereas  $\hat{G}_{11}$  has the unknown parameter  $\eta$ . However,  $\eta$  can be chosen arbitrarily (preferable according to nominal conditions) as the fuel flow is not measured and instead the linear gain from fuel valve stroke  $z_{fu}$  to fuel flow  $\dot{m}_{fu}$  can be estimated.

Regarding the feed water system the unknowns were the sensor time constant and the gain whether this is considered linear or produced by the solution to a quadratic equation.

All the unknown parameters can be identified by simple experiment such as steps in the fuel flow and staircase sequence in the feed water flow. Further these experiments can be performed during the boiler start up sequence not disturbing the availability of the boiler.

Of course, if other level sensors with less noise is available and a controller design based on a multivariable process model is used, more sophisticated experiments must be considered. This could be closed loop experiments to avoid disturbing normal operation too much having an initial PI controller installed and tuned as above.

## 5 CONCLUSION

In this paper we discussed performance limitations, system characteristics and simple control guidelines for the one-pass smoke tube marine boiler.

It was found that the measurement noise on the water level is what limits the achievable bandwidth. This led to the conclusion that benefits could be gained by a multivariable control structure as this allowed for speeding up the response from steam flow disturbance to water level through a disturbance estimate.

The control structure suggested was a cascade configuration where feedback and possible gain scheduling were applied to the feed water system whereas the fuel system was controlled by pure feedforward. The simple model used for the controller design makes controller auto-tuning relatively simple.

To improve performance it would be necessary to reduce the noise on the water level measurement. An opportunity could also be to use a differential pressure sensor to measure the amount of water in the boiler and control this instead of the actual level. Such a measurement is assumed to be less prone to noise. The idea is to combine this with a level setpoint controller which makes an estimate of the steam bubble volume to ensure that the actual water level does not violate the upper level constraints.

If other level measurements become available so that the bandwidth of the level loop can be moved closer to that of the pressure loop, then a multivariable control strategy should be applied to suppress the influence of interaction.

Given the hard constraints on the water level and the actuator limitations, MPC seems to be the natural choice from the control literature for marine boiler control. MPC has the advantage of allowing operation closer to the limits of the system, and further handle actuator constraints in a natural way. However, as the boiler only operates close to these limits when disturbances occur it seems reasonable to use another strategy and incorporate anti-windup to handle the few

cases when constraint bounds are active. Other advantages of MPC are the ease at which feedforward from the measured disturbance and future reference changes and disturbance changes can be incorporated in the design. However, neither of such information is available in case of the stand alone oil-fired marine boiler.

Instead it seems more appropriate to use a  $H_\infty$ /loop-shaping approach as such design methods have a natural way of including uncertainty and noise filters in the design through weight functions.

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